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(54) Variable displacement compressor and displacement control valve for variable displacement compressor

(57) A valve element 21 controlling refrigerant flow from a discharge chamber into a pressure-regulating chamber by reducing discharge pressure Pd to pressure Pc1, and a valve element 22 controlling refrigerant flow under pressure Pc2 from the pressure-regulating chamber into a suction chamber open and close in an interlocked fashion. A solenoid section applies solenoid force corresponding to a predetermined differential pressure valve to valve elements 21, 22. Either valve element 21 fully opens, while valve element 22 fully closes, or valve element 21 fully closes, and valve element 22 fully opens. Transitions between operating capacities can be performed rapidly. Valve element 21 is integral with a central shaft 25 for sensing pressure. Valve element 22 abuts the central shaft. The difference between the pressure-receiving areas of the valve elements 21, 22 and of the central shaft 25 is small to achieve a small effective pressure-receiving area for the valve elements 21, 22 to reduce the solenoid force for controlling the valve elements 21, 22.

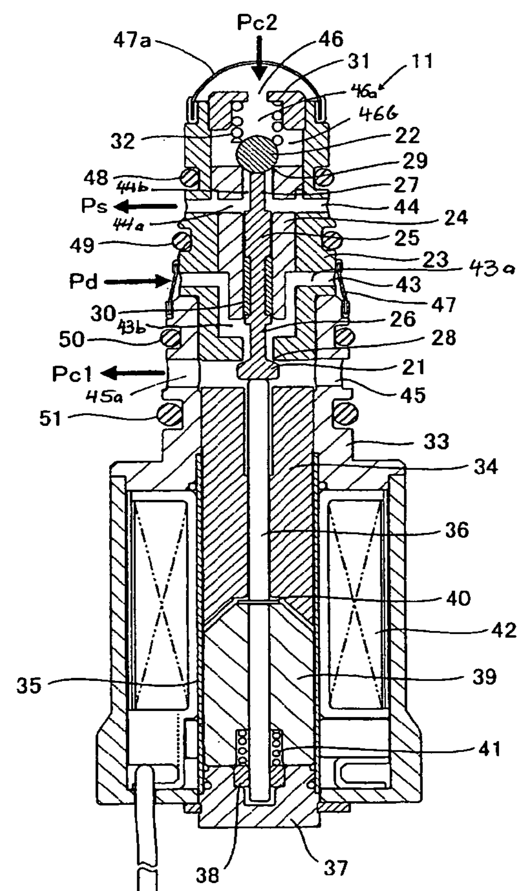

FIG. 2

EP 1 279 831 A2

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Description

[0001] This invention relates to a variable displacement compressor according to the preamble of claim 1 and to a displacement control valve according to the preamble part of claim 7.

[0002] A compressor in a refrigeration cycle for an automotive air conditioner is driven by the engine. The compressor speed cannot be controlled individually. For this reason, a variable displacement compressor is employed, the compression displacement of which is variable to obtain an adequate refrigerating displacement without dependence from the speed of the engine. In the compressor, pistons are connected to a wobble plate fitted on a shaft driven for rotation by the engine. The inclination angle of the wobble plate on the shaft is variable to change the piston stroke length for changing the delivery quantity.

[0003] The inclination angle of the wobble plate is continuously changed by introducing a part of compressed refrigerant into a pressure-regulating chamber and varying the pressure of the introduced refrigerant, thereby changing a balance between pressures applied to opposite sides of each piston.

[0004] A conventional compression displacement control device disclosed e.g. in JP 2001-132650 A has a solenoid control valve arranged between a discharge port and a pressure-regulating chamber or between the discharge port and a suction port of the compressor. This solenoid control valve opens and closes the communication such that the differential pressure across the solenoid control valve is maintained at a predetermined value set by a current value for the solenoid. When the engine rotational speed increases, the pressure introduced into the pressure-regulating chamber is increased to reduce the displacement for compression. When the engine rotational speed decreases, the pressure introduced into the pressure-regulating chamber is reduced to increase the displacement for compression. The pressure of refrigerant discharged from the compressor is maintained at a constant level.

[0005] The refrigerant used in a refrigeration cycle of an automotive air conditioner is a chlorofluorocarbon alternative (e.g. HFC-134a). Recently developed refrigeration cycles use refrigerants like carbon dioxide to perform refrigeration in a supercritical region where the temperature of the refrigerant is above its critical temperature.

[0006] The solenoid control valve has to maximize the amount of refrigerant introduced into the pressure-regulating chamber to minimize operating displacement. If the size of the valve is small, the amount or flow rate of refrigerant introduced is small, and the transition period to the minimum displacement operation takes a relatively long time, which degrades controllability of the compressor. If the size of the valve is increased to increase the amount of refrigerant introduced, the effective pressure-receiving area of the valve is also increased, and hence a large solenoid force is required to control the valve, i.e. to open it against the pressure acting on the pressure-receiving area. Particularly with carbon dioxide refrigerant the discharge pressure becomes very high, since it is increased to the supercritical region. Then the solenoid force controlling the valve needs to be large. A high solenoid force requires a huge solenoid which is heavy and costly and undesirably increases the size of the solenoid valve.

[0007] It is an object of the present invention to provide a variable displacement compressor and a displacement control valve for the variable displacement compressor which are capable of performing transitions between extreme operating capacities in a short time and do not need a large solenoid.

[0008] The present invention provides a variable displacement compressor, characterized in that a flow rate of the refrigerant flowing in a first refrigerant passage extending from the discharge chamber to the pressure-regulating chamber and a flow rate of the refrigerant flowing in a second refrigerant passage extending from the pressure-regulating chamber to the suction chamber are controlled in an interlocked fashion such that the first refrigerant passage and the second refrigerant passage are opened and closed, based on a change in a differential pressure between pressure in the suction chamber and pressure in the discharge chamber.

[0009] When controlling the operating displacement to the minimum, it is possible to open in an interlocking fashion the communication via the first refrigerant passage to the maximum and to close the second refrigerant passage, whereas in controlling the operating displacement to the maximum, it is possible to close in an interlocking fashion the first refrigerant passage and to open the communication via the second refrigerant passage to the maximum. As a result, when control to the minimum operating displacement is performed, introduction of refrigerant from the pressure-regulating chamber into the suction chamber is inhibited, and at the same time refrigerant is introduced at a maximum flow rate from the discharge chamber into the pressure-regulating chamber, whereas when control to the maximum operating displacement is performed, introduction of refrigerant from the discharge chamber into the pressure-regulating chamber is inhibited, and at the same time refrigerant is introduced at a maximum flow rate from the pressure-regulating chamber into the suction chamber. Therefore, time for transition to the minimum or maximum operating displacement can be shortened considerably.

[0010] To achieve a rapid transition e.g. to the minimum operating displacement by a compact solenoid with relatively small solenoid force, even with large sized valves, at least the valve closing the first passage is significantly pressure relieved with respect to the high discharge chamber pressure, such that the solenoid force only has to overcome a small differential force which is set independent from the actual valve size.

[0011] The displacement control valve controls the amount or flow rate of refrigerant introduced from a discharge chamber into a pressure-regulating chamber, such that a differential pressure between pressure in the suction chamber and pressure in the discharge chamber are maintained at a predetermined differential pressure value, to thereby change the amount of the refrigerant discharged from the variable displacement compressor, and is characterized by first and second valve elements operated in an interlocked fashion for opening and closing a refrigerant passage extending between the discharge chamber and the pressure-regulating chamber and a refrigerant passage extending between the pressure-regulating chamber and the suction chamber, respectively, and by a solenoid section for applying a solenoid force corresponding only to the predetermined differential pressure valve to the first and second valve elements.

[0012] The valve elements operate in an interlocked fashion, such that in controlling the operating displacement of the variable displacement compressor e.g. to the minimum, the second valve closes between the pressure-regulating chamber and the suction chamber, and the first valve opens to a maximum valve opening degree between the discharge chamber and the pressure-regulating chamber, whereas in controlling the operating displacement of the variable displacement compressor to the maximum, the second valve opens to a maximum valve opening degree between the pressure-regulating chamber and the suction chamber, and the first valve closes between the discharge chamber and the pressure-regulating chamber. When controlling to the minimum operating displacement, introduction of refrigerant from the pressure-regulating chamber into the suction chamber is inhibited, and at the same time refrigerant is introduced at a maximum flow rate from the discharge chamber into the pressure-regulating chamber, whereas when controlling to the maximum operating displacement, introduction of refrigerant from the discharge chamber into the pressure-regulating chamber is inhibited, and at the same time refrigerant is introduced at a maximum flow rate from the pressure-regulating chamber into the suction chamber. The time needed for the transition to the minimum or maximum operating displacement can be significantly shortened.

[0013] Large sized valves can be provided to achieve high maximum flow rates, when, e.g. like in one preferred embodiment, a central shaft is axially movably held by a holder fluidly separating the valve elements from each other and having a smaller pressure-receiving area than the valve elements provided on opposite ends of the central shaft to which the first valve element is fixed by an end shaft thinner than the central shaft, while another thin end shaft abuts at the second valve element. The discharge pressure from the discharge chamber is applied between the first valve element and the central shaft. The suction pressure from the suction chamber is applied between the second valve element and the central shaft. At the same time, a downstream side of the first valve element and a upstream side of the second valve element communicate with the pressure-regulating chamber by two independent passages.

[0014] Both valve elements receive on opposite sides the same pressure from the pressure-regulating chamber, whereby the influence of the pressure from the pressure-regulating chamber is cancelled. Only a small differential force (expressed by the product of the difference between the pressure-receiving areas of the first (second) valve element and the central shaft and a discharge (suction) pressure) acts on the first (second) valve element. Therefore, even if the size of each valve (of the valve element and the valve seat) is increased to increase the amount or flow rate of refrigerant allowed to flow when the operating displacement is to be changed, the respective active pressure-receiving areas of the first and second valve elements remain small, irrespective of the actual sizes of the valves, by reducing the difference between the pressure-receiving areas of the valve elements and the central shaft. Only a small solenoid force is needed to overcome the small differential force, which allows to downsize the solenoid section.

[0015] Embodiments of the invention will be described with reference to the drawings. In the drawings is:

Fig. 1 a cross-sectional view of a variable displacement compressor having a displacement control valve,

Fig. 2 a central longitudinal cross-sectional view of a first embodiment of the displacement control valve of Fig. 1,

Fig. 3 a cross-sectional view of a variable displacement compressor having another displacement control valve,

Fig. 4 a central longitudinal cross-sectional view of a second embodiment of the displacement control valve of Fig. 3,

Fig. 5 a central longitudinal cross-sectional view of a third embodiment of the displacement control valve,

Fig. 6 a cross-sectional view of a variable displacement compressor having another displacement control valve,

Fig. 7 a central longitudinal cross-sectional view of a displacement control valve of Fig. 6, and of the fourth embodiment, and

Fig. 8 a central longitudinal cross-sectional view of a fifth embodiment of the displacement control

valve.

[0016] The variable displacement compressor of Fig. 1, e.g. of a refrigeration cycle of an automotive air-conditioning system, includes an airtight pressure-regulating chamber 1 and a rotatably supported rotational shaft 2 one end of which carries a pulley 3 connected with an output shaft of an engine via a clutch and a belt. A wobble plate 4 is fitted on the rotational shaft 2 such that the inclination angle of the wobble plate 4 can be changed. A plurality of cylinders 5 (only one of which is shown in the figure) are arranged around the axis of the rotational shaft 2. Each cylinder 5, receives a piston 6 and is connected to a suction chamber 9 and a discharge chamber 10 via a suction relief valve 7 and a discharge relief valve 8, respectively. The suction chambers 9 communicate with each other to form one chamber which is connected to an evaporator. The discharge chambers 10 communicate with each other to form one chamber which is connected to a gas cooler or a condenser.

[0017] A displacement control valve 11 comprising two valves is arranged at intermediate portions of a refrigerant passage 10a, 10b extending from the discharge chamber 10 to the pressure-regulating chamber 1 and a refrigerant passage 1a, 1b between the pressure-regulating chamber 1 and the suction chamber 9. There are formed orifices 12, 13 between the discharge chamber 10 and the pressure-regulating chamber 1 and between the pressure-regulating chamber 1 and the suction chamber 9, respectively. The orifices 12, 13 may be formed in the compressor body or in the displacement control valve 11.

[0018] Rotation of the wobble plate 4 causes reciprocating motions of the pistons 6. Refrigerant within the suction chamber 9 is drawn into the cylinder 5, and is compressed and then delivered to the discharge chamber 10.

[0019] During normal operation, responsive to a discharge pressure Pd within the discharge chamber 10, the displacement control valve 11 controls the amount of refrigerant introduced into the pressure-regulating chamber 1 (the pressure in the pressure-regulating chamber 1 at the time is represented by Pc1) and the amount of refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9 (the pressure in the pressure-regulating chamber 1 at the time is represented by Pc2) in an interlocking fashion, such that a predetermined differential pressure value is maintained between the discharge pressure Pd and a suction pressure Ps. The pressure Pc (= Pc1 = Pc2) in the pressure-regulating chamber 1 is held at a predetermined value, and the displacement in the cylinder 5 is controlled to a predetermined value.

[0020] For performing transition to a minimum displacement operation the displacement control valve 11 fully opens one of its two valves which serve to introduce refrigerant from the discharge chamber 10 into the pressure-regulating chamber 1 and fully closes the other valve which serves to introduce refrigerant from the pressure-regulating chamber 1 into the suction chamber 9. By this action the time for increasing the pressure Pc (= Pc1) in the pressure-regulating chamber 1 is shortened. Although then the displacement control valve 11 fully closes the refrigerant passage extending from the pressure-regulating chamber 1 to the suction chamber 9, a minute flow will take place via orifice 13.

[0021] To adjust the maximum displacement operation the displacement control valve 11 fully closes the one valve provided for introducing refrigerant from the discharge chamber 10 into the pressure-regulating chamber 1 and fully opens the other valve provided for introducing refrigerant from the pressure-regulating chamber 1 into the suction chamber 9, so as to maximize the amount or flow rate of refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9. This action shortens the time for reducing the pressure Pc (= Pc2) in the pressure-regulating chamber 1. Although the displacement control valve 11 fully closes the refrigerant passage 10a, 10b still some refrigerant passes the orifice 12, such that lubricating oil mixed into the refrigerant is supplied to the pressure-regulating chamber 1.

[0022] The first embodiment of the displacement control valve 11 of Fig. 2 comprises two integrally formed valve elements 21, 22 operating in an interlocked fashion, i.e. they are coupled to each other by a form-fit when alternatingly opening or closing their respective valve seats. A central shaft 25 is axially guided by a holder 24 in a central opening portion of a body 23. Shaft 25 has thinned end shafts 26, 27. Valve element 21 is integrally formed with the end shaft 26. Valve element 22 is in abutment with the upper thin end shaft 27. The central shaft 25 has a pressure-receiving area smaller than the pressure-receiving areas of the valve elements 21, 22 and forms a pressure-sensing portion. Further, central shaft 25 is formed with a reduced diameter portion between both end shafts 25, 26, on which a packing 30 e. g. of polytetrafluoroethylene, is fitted.

[0023] A valve seat 28 for the valve element 21 is formed by the lower end of the body 23. The valve seat 28 is the mouth of a valve hole of an inner diameter slightly larger than the inner diameter of the holder 24.

[0024] A valve seat 29 for the valve element 22 directly is formed by the upper end of an inner bore of the holder 24. The valve seat 29 is the mouth of a valve hole of an inner diameter slightly larger than the inner diameter of the bore portion of the holder 24 holding the central shaft 25. The valve element 22 is biased in valve-closing direction by a spring 32 abutting at a spring-receiving member 31.

[0025] The body 23 is fitted in an upper opening of a body 33. The body 33 contains a fixed core 34 and an outer sleeve 35 of a solenoid section. The fixed core 34 has a central opening portion axially guiding a shaft 36. The lower end of the shaft 36 is axially guided by a guide 38 arranged in a stopper 37 closing the lower end of the

sleeve 35. A movable core 39 is fitted on the lower portion of the shaft 36. A movable core upper end is held in abutment at a stopper ring 40 fitted on the shaft 36, and is urged upward by a spring 41 abutting at the guide 38. The sleeve 35 is surrounded by a solenoid coil 42.

[0026] The body 23 has a lateral hole 43a communicating with a central space 43b through which the thin end shaft 26 extends. The hole 43a forms at least one port 43 receiving discharge pressure Pd from the discharge chamber 10. A strainer 47 covers the port 43. The body 23 has a further hole 44a communicating with a central space 44b through which the thin end shaft 27 extends. The hole 44a forms at least one port 44 receiving suction pressure Ps from the suction chamber 9. The body 33 has a hole 45a communicating with a space 45b in which the valve element 21 is arranged. The hole forms a port 45 for pressure Pc1 to the pressure-regulating chamber 1. The spring-receiving member 31 has an axial hole 46a communicating with a space 64b in which the valve element 22 is arranged. The hole 46a forms a port 46 for pressure Pc2 from the pressure-regulating chamber 1. A strainer 47a is mounted on a distal end of the body 23. The body 23 has O-rings 48, 49 at locations above and below the port 44, while the body 33 has O-rings 50, 51 at locations above and below the port 45.

[0027] The discharge pressure Pd from the discharge chamber 10 acts via port 43 on the central shaft 25 and on the valve element 21 in opposite axial directions. When the effective pressure-receiving area of the valve element 21 is represented by A, and that of the central shaft 25 by B, a force of Pd · A acts downward, on the valve element 21, while a force of Pd · B acts upward, as viewed in the figure, on the central shaft 25. Between the effective pressure-receiving area A of the valve element 21 and the effective pressure-receiving area B of the central shaft 25, A > B holds, and hence, after all, a force of Pd (A — B) acts on the valve element 21 and the central shaft 25 in the downward direction, as viewed in the figure, for opening the valve. The area difference (A — B) e.g. may correspond to the effective pressure-receiving area of the conventional valve element, and conventionally, the possible flow rate of refrigerant is limited by nature by the effective pressure-receiving area. According to the present invention, however, although the large sized valve element 21 has a large effective pressure-receiving area A allowing an increased amount or flow rate of refrigerant, the force acting on the valve element 21 in the valve-opening direction is limited to the small differential force Pd (A — B). Further, since the pressures Pc1, Pc2 (Pc1 = Pc2) in the pressure-regulating chamber 1 are axially applied to the valve elements 21, 22 from respective opposite sides via the respective ports 45, 46, the influence of the pressure Pc upon the valve element 21 is canceled. Thus, the central shaft 25 has a different pressure-receiving area than the integrally formed valve element 21. This makes it possible to provide a valve having the small pressure-receiving area of (A — B), irrespective of the actual valve size, always provided that area B is somewhat smaller than area A.

[0028] The force of Ps (A — B) acts on the valve element 22 and the central shaft 25 in the valve-opening direction. The pressures Pc1, Pc2 (Pc1 = Pc2) are axially applied to the valve elements 21, 22 integral with each other (coupled by central shaft 25) from the opposite sides, which cancels the influence of the pressure Pc upon the valve element 22. The ratio between the effective pressure-receiving areas of the valve element 22 and the central shaft 25 is set equal to the ratio between the effective pressure-receiving areas of the valve element 21 and the central shaft 25. As a result, both valve elements 21, 22 constitute a differential pressure valve operating in response to a differential pressure between the discharge pressure Pd and the suction pressure Ps.

[0029] The pressure Pc1 at port 45 is supplied to a gap between the sleeve 35 and the movable core 39 and to a gap between the movable core 39 and the stopper 37 via a clearance between the fixed core 34 and the shaft 36, such that the inside of the solenoid section contains the pressure Pc1.

[0030] As long as no control current is supplied to the solenoid coil 42 (FIG. 2) the valve element 21 fully opens valve seat 28, whereas the valve element 22 fully closes valve seat 29. The movable core 39 stays away from the fixed core 34 due to a balance between the spring loads of springs 32, 41. The value of the pressure Pc1 in the pressure-regulating chamber 1 is held close to the valve of the discharge pressure Pd. The difference between pressures acting at the opposite faces of each piston 6 is minimized. The wobble plate 4 is inclined by an inclination angle which minimizes the piston stroke length. The variable displacement compressor is controlled to the minimum displacement operation.

[0031] As soon as maximum control current is supplied to the solenoid coil 42, the movable core 39 is attracted toward the fixed core 34 and moves upward. The valve element 21 fully closes valve seat 28, and the valve element 22 fully opens valve seat 29. In addition to refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9 via the orifice 13, refrigerant will flow from port 46 through valve seat 29 and will be introduced into the suction chamber 9 via the port 44. Since the amount or flow rate of refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9 is increased, it is possible to increase the speed or to shorten the transition time until the operating displacement is maximized.

[0032] During normal control with a predetermined control current for the solenoid coil 42, the movable core 39 is attracted toward the fixed core 34 and moved upward. according to the magnitude of the control current. The valve element 22 clears valve seat 28 only when the differential pressure between the discharge pressure Pd and the suction pressure Ps exceeds a prede-

termined reference value. In short, during normal control, the displacement control valve 11 is operating as a differential pressure valve.

[0033] In the variable displacement compressor of Fig. 3, which is structured similar to the one in Fig. 1, a second embodiment of the displacement control valve 60 comprising two valves is arranged as in Fig. 1. Normal operation is performed as described in connection with Fig. 1 The transition to the minimum displacement operation is performed, as described in connection with Fig. 1. The adjustment to a maximum displacement operation of the compressor is carried out as described for Fig. 1.

[0034] In the second embodiment of the displacement control valve 60 of Fig. 4, two valve elements 61, 62 are located opposite to each other with a transmission shaft 63 arranged between them. The valve elements 61, 62 move along a common axis. The two valve structures are interlocked by the transmission shaft 63. The valve element 61, as viewed in the figure, is integrally upper formed with a piston 64 forming a pressure-sensing portion, a shaft 65 connecting the valve element 61 and the piston 64, and an axial communication hole 66. Similarly, the lower valve element 62 is integrally formed with a piston 67 forming a pressure-sensing portion, a shaft 68 between the valve element 62 and the piston 67, and an axial inner communication hole 69. Each valve element 61, 62 has an end face in abutment with the transmission shaft 63. The respective end face is formed with a step (not shown in detail) allowing communication despite the abutment between the communication hole 66 (69) and a space where the valve element 61 (62) is located.

[0035] A valve seat 70 for the valve element 61 is formed by the lower end, as viewed in the figure, of a central cylinder bore of a body 71 axially guiding the piston 64. The valve seat 70 has an inner diameter slightly larger than the inner diameter of the cylinder bore guiding the piston 64. The valve element 61 is biased in valve-opening direction by a spring 72.

[0036] The body 71 is fitted into an upper opening of a body 73 having a central hole extending downward from the upper opening. The hole has four stepwise sequentially reduced-diameter portions. A first reduced-diameter portion receives a holder 74 axially guiding the transmission shaft 63. An opening mouth of a step to a next reduced-diameter portion forms a valve seat 75 for the valve element 62. A next reduced-diameter portion forms a cylinder bore for axially guiding the piston 67. A next reduced-diameter portion forms a guide for axially slidably holding a shaft 76 of a solenoid section. The lower portion of the body 73 forms a fixed core 78.

[0037] The body 73 is screwed into an upper opening of a body 79. The upper end of a sleeve 80 closed by a stopper 81 is fixed to a lower opening of the body 79. Within sleeve 80, the lower end of the shaft 76 is axially guided by a guide 82 provided in the stopper 81. A movable core 83 is fitted on the lower portion of the shaft 76. The movable core 83 has an upper end for abutment with a stopper ring 84 fitted on the shaft 76, and is urged upward, as viewed in the figure, by a spring 85 abutting at the guide 82. The sleeve 80 is surrounded by a solenoid coil 86.

[0038] The body 71 has a hole communicating with a central space through which the shaft 65 extends. The hole forms a port 87 for the discharge pressure Pd. A strainer 88 covers the port 87. The body 73 has a hole communicating with a space receiving the valve element 61. The hole forms a port 89 for pressure Pc1 to the pressure-regulating chamber 1. The body 73 has a further lateral hole communicating with a space receiving the valve element 62. The hole forms a port 90 for introducing pressure Pc2. The body 73 further has a lateral hole for communication with a central space through which the shaft 68 extends. The body 79 is formed with a lateral hole communicating with the hole of the body 73. Both holes define a port 91 communicating with the suction chamber 9 under the suction pressure Ps.

[0039] The body 73 has O-rings 92, 93 above and below the port 89. The body 79 has O-rings 94, 95 above and below the port 91. Contacting portions of the bodies 73, 79 close to the solenoid section are sealed with respect to the port 91 by an O-ring 96.

[0040] The discharge pressure Pd at port 87 is applied to the piston 64 and the valve element 61 in opposite axial directions. When the effective pressure-receiving area of the valve element 61 is represented by A, and that of the piston 64 by B, a force of Pd · A acts downward, as viewed in the figure, on the valve element 61, while a force of Pd · B acts upward on the piston 64. Between the effective pressure-receiving areas A, B of the valve element 61 and the piston 64, A > B holds, and hence, a force of Pd (A — B) acts on the valve element 61 and the piston 64 in the downward direction for opening the valve. The area difference (A — B) e.g. corresponds to the effective pressure-receiving area of the conventional valve element. According to the present invention, however, although the large sized valve element 61 has a large effective pressure-receiving area A allowing an increased amount or flow rate of refrigerant Pd acting on the valve element 61 in the valve-opening direction generates only the small differential force Pd (A — B). The pressure Pc1 at port 89 is applied to a back pressure chamber-side face of the piston 64 via the central communication hole 66, so that the influence of the pressure Pc1 upon the valve element 61 is canceled. Thus, the piston 64 integrally formed with the valve element 61 has a different pressure-receiving area than the valve element 61. This makes it possible to provide a valve having a small pressure-receiving area, irrespective of the actual valve size.

[0041] Similarly, a force of Ps (A — B) acts on the valve element 62 and the piston 67 in the valve-opening direction. The pressure Pc2 at port 90 is applied to a back pressure chamber-side face of the piston 67 via

the central communication hole 69, so that the influence of the pressure Pc2 upon the valve element 62 is canceled. The ratio between the effective pressure-receiving areas of the valve element 62 and the piston 67 is set to be equal to the ratio between the effective pressure-receiving areas of the valve element 61 and the piston 64. Therefore, the valve elements 61, 62 in opposed arrangement form a differential pressure valve operating in response to a differential pressure between the discharge pressure Pd and the suction pressure Ps.

[0042] The pressure Pc2 at port 90 is supplied via the communication hole 69 to a space forming the back-pressure chamber of the piston 67, and passes through a clearance between the fixed core 78 and the shaft 76, a space between the fixed core 78 and the movable core 83, a clearance between the sleeve 80 and the movable core 83, and a clearance between the movable core 83 and the stopper 81, i.e. the internal part of the displacement control valve 60 closer to the solenoid section with respect to the O-ring 96 contains the pressure Pc2 (= Pc).

[0043] As long as no control current is supplied to the solenoid coil 86 (FIG. 4), the valve element 61 fully opens valve seat 70, whereas the valve element 62 fully closes valve seat 75. The movable core 83 stays away from the fixed core 78 due to a balance between the spring loads of spring 72, 85. The value of the pressure Pc1 in the pressure-regulating chamber 1 is held close to the value of the discharge pressure Pd. The difference between the pressures applied to the respective opposite faces of the pistons 6 is minimized. The wobble plate 4 has an inclination angle which minimizes the piston stroke length. The variable displacement compressor performs the minimum displacement operation.

[0044] As soon as a maximum control current is supplied to the solenoid coil 86 the movable core 83 is attracted toward the fixed core 78 and moved upward. The valve element 61 fully close valve seat 70" the valve element 62 fully opens the valve seat 75. In addition to refrigerant flowing from the pressure-regulating chamber 1 into the suction chamber 9 via the orifice 13, refrigerant flows from port 90 through valve seat 75 and into the suction chamber 9 via port 91. Since the amount or flow rate of refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9 is increased, it is possible to increase the speed or shorten the time until the operating displacement is maximized.

[0045] During normal control by a predetermined control current supplied to the solenoid coil 86, the movable core 83 is attracted toward the fixed core 78 and moved upward, as viewed in the figure, however only according to the magnitude of the control current. The valve element 62 opens valve seat 75 only when the differential pressure between the discharge pressure Pd and the suction pressure Ps exceeds a predetermined reference value. During normal control, the displacement control valve 60 operates as a differential pressure valve.

[0046] The displacement control valve 60a according to the third embodiment of Fig. 5 has a different structure for canceling the influences of the pressures Pc1, Pc2 on the valve elements 61, 62. More specifically, a one-piece member formed by the valve element 61, a piston 64, and a shaft 65, and a one-piece member formed by the valve element 62, a piston 67, and a shaft 68 are each formed as a solid member without axial communication holes. In this case, the body 71 is formed with a communication hole 97 for introducing the pressure Pc1 into a back pressure chamber of the piston 64. The body 73 is formed with a communication hole 77 opening into a space forming a back pressure chamber of the piston 67 and forms a clearance between respective portions of fixed core 78 and sleeve 80 close to the solenoid section with respect to O-ring 96. The displacement control valve 60a of Fig. 5 operates similarly to the displacement control valve 60 of Fig. 4.

[0047] In the variable displacement compressor of Fig. 6 the fourth embodiment of the displacement control valve 100 comprises two valves and is arranged at an intermediate portion of a refrigerant passage 10a, 10b extending from the discharge chamber 10 to the pressure-regulating chamber 1 and in a refrigerant passage 1a, 1b communicating the pressure-regulating chamber 1 and the suction chamber 9. The two refrigerant passages 10a, 10b, 1a, 1b share the passage portion 10b, 1a between the displacement control valve 100 and the pressure-regulating chamber 1.

[0048] During normal operation, responsive to discharge pressure Pd of the refrigerant within the discharge chamber 10, the displacement control valve 100 controls the amount or flow rate of refrigerant introduced into the pressure-regulating chamber 1 and the amount of refrigerant which is part of refrigerant to be introduced into the pressure-regulating chamber 1 but is supplied into the suction chamber 9 in a bypassing manner, such that the differential pressure between the discharge pressure Pd and a suction pressure Ps is maintained at a predetermined differential pressure value. As a result, the pressure Pc in the pressure-regulating chamber 1 is held at a predetermined value, and the displacement of the cylinders 5 is controlled to a predetermined value. Thereafter, the pressure Pc in the pressure-regulating chamber 1 is returned to the suction chamber 9 via orifice 13.

[0049] The transition to the minimum displacement operation is performed as described for Fig. 1.

[0050] The transition to the maximum displacement operation is performed, rapidly as described for Fig. 1.

[0051] In the fourth embodiment of the displacement control valve 100 of Fig. 7, the two valve elements 101, 102 are moveably arranged directly opposed to each other such that they move along a common axis. The upper valve element 101, as viewed in the figure, is integrally formed with a piston 103 forming a pressure-sensing portion, a shaft 104 between the valve element 101 and the piston 103 and an axial communication hole

105. Similarly, the lower valve element 102 is integrally formed with a piston 106 forming a pressure-sensing portion, a shaft 107 between the valve element 102 and the piston 106. and an axial communication hole 108. The valve elements 101, 102 have end faces abutting each other, and each formed with a step (not shown in detail) allowing communication between the communication hole 105 (108) and a space where the valve element 101 (102) is located.

[0052] A valve seat 109 for the valve element 101 is formed by the lower end of a body 110 axially guiding the piston 103. The valve seat inner diameter is slightly larger than the diameter of the piston 103. The valve element 101 is biased in valve-opening direction by a spring 112 arranged between an E-shaped stopper ring 111 on the valve element 101 and the body 110.

[0053] The body 110 is fitted in an upper opening of a body 113. The body 113 has a bore having three stepwise sequentially reduced-diameter portions. An opening mouth in a step to a first reduced-diameter portion forms a valve seat 114 for the valve element 102. The next reduced-diameter portion defines a cylinder bore for axially guiding the piston 106. The next reduced-diameter portion forms a guide bore axially guiding a shaft 115. The body 113 has a communication hole 116 parallel to the axis. The lower end of the communication hole 116 communicates with a lateral communication hole between the guide bore of the shaft 115 and the outer periphery of the body 113. The lower portion of the body 113 forms the fixed core 117 of the solenoid section.

[0054] The body 113 is threaded into the upper opening of a body 118. The upper end of a sleeve 119 closed by a stopper 120 is fixed in a lower opening of the body 118. In sleeve 119, the lower end of the shaft 115 is axially guided by a guide 121. A movable core 122 is fitted on the lower portion of the shaft 115. The movable core 122 has an upper end for abutment with a stopper ring 123 fitted on the shaft 115, and is urged upward, as viewed in the figure, by a spring 124 abutting at the guide 121. The sleeve 119 is surrounded by a solenoid coil 125.

[0055] The body 110 has a lateral hole communicating with a central space through which the shaft 104 extends, and the hole forms a port 126 for the discharge pressure Pd. A strainer 127 covers the port 126. The body 113 has a lateral hole communicating with a central space formed in the upper opening portion thereof, and the hole forms a port 128 for introducing pressure Pc into the pressure-regulating chamber 1. Further, the body 113 has a lateral hole communicating with a central space through which the shaft 107 extends. The body 118 has a lateral hole communicating with the hole of the body 113 and forming a port 129 communicating with the suction chamber 9.

[0056] The body 113 has an O-ring 130 between the port 126, 128, while the body 118 has O-rings 131, 132 above and below the port 129. Contacting portions of the body 113 and the body 118 close to the solenoid section with respect to the port 129 are sealed by an O-ring 133.

[0057] The discharge pressure Pd at port 126 is applied to the piston 103 and the valve element 101 in opposite axial directions. When the effective pressure-receiving area of the valve element 101 is represented by A, and that of the piston 103 by B, a force of Pd · A acts downward, as viewed in the figure, on the valve element 101, while a force of Pd · B acts upward, as viewed in the figure, on the piston 103. Between the effective pressure-receiving area A of the valve element 101 and the effective pressure-receiving area B of the piston 103, A > B holds, and hence, after all, a force of Pd (A — B) acts on the valve element 101 and the piston 103 in the downward direction, as viewed in the figure, for opening the valve. The difference area (A — B) corresponds e. g. to the entire effective pressure-receiving area of the conventional valve element. According to the present invention, however, although the large sized valve element 101 has the large effective pressure-receiving area A which when fully opened allows an increased amount or flow rate of refrigerant, discharge pressure Pd acts on the valve element 101 in the valve-opening direction only with a limited small differential force Pd (A — B). Moreover, pressure Pc at port 128 is applied to a back pressure chamber-side face of the piston 103 via the central communication hole 105, so that the influence of the pressure Pc upon the valve element 101 is canceled. The piston 103 integrally formed with the valve element 101 has a different pressure-receiving area than the valve element 101. This makes it possible to provide a valve operating with a small pressure-receiving area, irrespective of the actual valve size.

[0058] Similarly, the differential force Ps. (A — B) acts on the valve element 102 and the piston 106 in valve-opening direction. The pressure Pc at port 128 is applied to a back pressure chamber-side face of the piston 106 via the central communication hole 108, so that the influence of the pressure Pc upon the valve element 102 is canceled. The ratio between the effective pressure-receiving areas of the valve element 102 and the piston 106 is set to be equal to the ratio between the effective pressure-receiving areas of the valve element 101 and the piston 103. Therefore, the valve elements 101, 102 in opposed arrangement commonly operate as a differential pressure valve in response to a differential pressure between the discharge pressure Pd and the suction pressure Ps.

[0059] The pressure Pc at port 128 is supplied via communication hole 116 to a gap between the sleeve 119 and the fixed core 117 and the movable core 122, a space between the fixed core 117 and the movable core 122, and a gap between the movable core 122 and the stopper 120, so that the inside of the solenoid section contains the pressure Pc.

[0060] As long as no control current is supplied to the solenoid coil 125 in FIG. 7, the valve element 101 fully

opens valve seat 109, whereas the valve element 102 fully closes valve seat 114. The movable core 122 stays away from the fixed core 117 due to a balance between spring loads of the springs 112, 124. The value of the pressure Pc in the pressure-regulating chamber 1 is held close to the value of the discharge pressure Pd, and hence the difference between pressures applied to the respective opposite faces of the pistons 6 is minimized. The wobble plate 4 has an inclination angle which minimizes the piston stroke length for controlling the minimum displacement operation.

[0061] As soon as a maximum control current is supplied to the solenoid coil 125, the movable core 122 is attracted toward the fixed core 117 and moved upward, as viewed in the figure, until the valve element 101 fully closes valve seat 109, and the valve element 102 fully opens valve seat 114. In addition to refrigerant leaking from the pressure-regulating chamber 1 into the suction chamber 9 via the orifice 13, refrigerant will flow from port 128 through the valve seat 114 and via the port 129 into the suction chamber 9. Since the amount of refrigerant introduced from the pressure-regulating chamber 1 into the suction chamber 9 is increased, it is possible to increase the speed at which the operating displacement is maximized.

[0062] During normal control by a predetermined control current supplied to the solenoid coil 125, the movable core 122 is attracted toward the fixed core 117 and moved upward, with a force according to the magnitude of the control current. The valve element 102 clears valve seat 114 only when the differential pressure between the discharge pressure Pd and the suction pressure Ps exceeds a predetermined reference value. In short, during normal control, the displacement control valve 100 operates as a differential pressure valve.

[0063] The fifth embodiment of the displacement control valve 100a of Fig. 8 has a different structure for canceling the influence of the pressure Pc upon valve elements 101, 102. More specifically, the one-piece members formed by the valve element 101, the piston 103 and the shaft 104, and by the valve element 102, the piston 103 and the shaft 107 are each formed without axial communication holes. In this case, the body 110 has a communication hole 134 for introducing the pressure Pc into a back pressure chamber of the piston 103. Further, the body 113 is formed with a communication hole 116 opening into a space forming a back pressure chamber of the piston 106. The body 113 has a gap between respective portions of the fixed core 117 and the sleeve 119 close to the solenoid section with respect to O-ring 133. The displacement control valve 100a operates similarly to the displacement control valve 100 of the fourth embodiment.

Claims

1. A variable displacement compressor including a wobble member (4) arranged in an airtight pressure-regulating chamber (1), such that an inclination angle of the wobble member (4) can be changed with respect to a rotational shaft (2), driving the wobble member for a wobbling motion, and pistons (6) connected to the wobble member for performing reciprocating motions in directions parallel to the rotational shaft in accordance with the wobbling motion of the wobble member, to thereby draw refrigerant from a suction chamber (9) into a cylinder (5), compress the refrigerant, and deliver the compressed refrigerant from the cylinder to a discharge chamber (10),
   characterized in that a flow rate of the refrigerant flowing in a first refrigerant passage (10a, 10b) extending from the discharge chamber (10) to the pressure-regulating chamber (1) and a flow rate of the refrigerant flowing in a second refrigerant passage (1a, 1b) extending from the pressure-regulating chamber (1) to the suction chamber (9) are controlled in an interlocked fashion such that the first refrigerant passage and the second refrigerant passage are opened and closed, based on a change in a differential pressure between pressure (Ps) in the suction chamber (9) and pressure (Pd) in the discharge chamber (10).

2. The variable displacement compressor as in claim 1, **characterized in that** the first refrigerant passage extends in parallel with a first orifice (12) for introducing the refrigerant from the discharge chamber (10) into the pressure-regulating chamber (1), while the second refrigerant passage extends in parallel with a second orifice (13) for introducing the refrigerant from the pressure-regulating chamber (1) to the suction chamber (9).

3. The variable displacement compressor as in claim 1, **characterized in that** when the compressor is operated with a minimum operating displacement, the first refrigerant passage is fully opened, and the second refrigerant passage fully closed, whereas when the compressor is operated with a maximum operating displacement, the first refrigerant passage is fully closed, and the second refrigerant passage fully opened.

4. The variable displacement compressor as in claim 1, **characterized by** a solenoid controlled displacement control valve (11; 60; 60a; 100; 100a) comprising a first valve (21, 28; 61, 70; 101, 109) in the first passage (10a, 10b) and a second valve (22, 29; 62, 75; 102, 114) in the second passage (1, 1a, 1b), each valve consisting of a valve element and a valve seat,
   both valve elements being mechanically interlocked for common alternating opening and closing travels relative to the associated valve seat,

the valve element (21, 61, 101) of the first valve being loaded on an effective pressure receiving area (A) by the discharge chamber pressure (Pd) in valve opening direction,
at least the valve element (21, 61, 101) of the first valve being integrally connected to a pressure responsive piston-like component (25; 64; 103) guided in sealed fashion for common travel with the valve element,
the component defining an effective discharge chamber pressure receiving area (B) only slightly smaller than the effective discharge chamber pressure receiving area (A) of the valve element (21; 61; 101),
the pressure responsive component being loaded by the discharge chamber pressure (Pd) in valve closing direction of the valve element,
the solenoid force generated by current supplied to a solenoid section acting on the valve element of the first valve in valve closing direction counter to at least a differential force generated by the discharge chamber pressure (Pd) on the effective pressure receiving areas difference (A — B).

5. The variable displacement compressor as in claim 4, **characterized in that** the valve element (22; 62; 102) of the second valve is actuable by a pressure responsive piston-like component (25; 67; 106) guided in sealed fashion for common travel with the valve element,
the pressure responsive piston-like component defining an effective pressure receiving area (B) only slightly smaller than the effective pressure receiving area (A) of the valve element (22; 62; 102),
the pressure responsive component (25; 67; 106) being loaded by the suction chamber pressure (Ps) in valve closing direction of the valve element.

6. The variable displacement compressor as in claims 4 and 5, **characterized in that** both respective ratios (A:B) between the effective pressure receiving area of each valve element and the associated piston-like component are equal.

7. A displacement control valve (11, 60, 60a, 100, 100a) for a variable displacement compressor, for controlling an amount of refrigerant introduced from a discharge chamber (10) into a pressure-regulating chamber (1), such that a differential pressure between pressure in a suction chamber (9) and pressure in the discharge chamber (10) is maintained at a predetermined differential pressure value, to thereby change an amount of the refrigerant discharged from the variable displacement compressor,
characterized by:

first and second valve elements (21, 22; 61, 62; 101, 102) operated in an interlocked fashion for alternatingly opening and closing valve seats (28, 29; 70, 75; 100, 114) in a refrigerant passage extending between the discharge chamber (10) and the pressure-regulating chamber (1) and a refrigerant passage extending between the pressure-regulating chamber (1) and the suction chamber (9), respectively; and
a solenoid section for commonly applying a solenoid force corresponding to the predetermined differential pressure to the first and second valve elements (21, 22; 61, 62; 101, 102).

8. The displacement control valve as in claim 7, **characterized in that** a central shaft (25) axially movably held by a holder (24) fluidly separating the first valve element (21) and the second valve element (22) from each other and having a smaller pressure-receiving area (B) than the pressure receiving areas (A) of the first and second valve elements has opposite ends one of which has the first valve element (21) fixed thereto via an end shaft (26) thinner than the central shaft (25) and the other of which has the second valve element (22) in abutment therewith via an end shaft (27) thinner than the central shaft (25), with discharge pressure (Pd) from the discharge chamber (10) being applied between the first valve element (21) and the central shaft (25), and suction pressure (Ps) from the suction chamber (9) being applied between the second valve element (22) and the central shaft (25), and wherein at the same time, a downstream side of the first valve element (21) and a upstream side of the second valve element (22) are communicated with the pressure-regulating chamber (1) by respective two passages independent of each other.

9. The displacement control valve as in claim 8, **characterized in that** the solenoid section is arranged at the side of the first valve element (21), and a shaft (36) of the solenoid section for applying the solenoid force to the first valve element (21) is in abutment with the first valve element (21).

10. The displacement control valve as in claim 9, **characterized in that** the first valve element (21), the central shaft (25), and both thinner end shafts (26, 27) of the central shaft (25), are integrally formed with each other, the second valve element (22) being held in abutment at the end shaft (27) at the opposite side of the first valve element (21).

11. The displacement control valve as in claim 7, **characterized in that** the first and second valve elements (61, 62) are arranged opposed to each other on an identical axis in an axially movable fashion, that each valve element (61, 62) is integrally formed with an axial shaft (65, 68) and an axially outwardly

located piston (64, 67) having a smaller pressure-receiving area (B) than the valve element, that a holder (74) fluidly separates the valve elements from each other, and that a transmission shaft (63) is sandwiched between the valve elements for axially moving the valve elements in an interlocking fashion, axially slidably guided by the holder (74), that discharge pressure (Pd) from the discharge chamber (10) is applied between the first valve element (61) and the piston (64), and suction pressure (Ps) from the suction chamber (9) is applied between the second valve element (62) and the piston (67), and that at the same time, a downstream side of the first valve element (61) and a upstream side of the second valve element (62) communicate with the pressure-regulating chamber (1) by two independent passages.

12. The displacement control valve as in claim 8, **characterized in that** a first communication hole (66; 97) extends between a back pressure chamber of the piston (64) and the downstream side of the first valve element (61), for introducing pressure (Pc1) from the pressure-regulating chamber (1), and that a second communication hole (69; 77) extends between a back pressure chamber of the piston (67) and the upstream side of the second valve element (62), for introducing pressure (Pc2) from the pressure-regulating chamber (1).

13. The displacement control valve as in claim 12, **characterized in that** the first communication hole (66) extends along the axis of the first valve element (61), the shaft (65), and the piston (64), integrally formed with each other, that the second communication hole (69) extends along the axis of the second valve element (62), the shaft (68), and the piston (67), integrally formed with each other, and that portions of the valve elements (61, 62) abutting at the transmission shaft (63) communicate with the first and second communication holes (66, 69), respectively, preferably via stepped abutment faces of the respective valve element (61, 62) and/or of the transmission shaft (63).

14. The displacement control valve as in claim 12, **characterized in that** the first communication hole (97) extends through a first body (71) providing a valve seat (70) for the first valve element (61), and that the second communication hole (77) extends through a second body (73) providing a valve seat (75) for the second valve element (62).

15. The displacement control valve as in claim 11, **characterized in that** a body (73) providing a valve seat (75) for the second valve element (62) and defining a cylinder bore slidably guiding the piston (67) is integrally formed with a fixed core (78) of the solenoid section.

16. The displacement control valve as in claim 7, **characterized in that** the valve elements (101, 102) are commonly movably arranged opposed to each other on a common axis and directly abut each other, that each valve element (101, 102) is integrally formed with an axial shaft (104, 107) and an outwardly located piston (103, 106) having a smaller pressure-receiving area (B) than the valve element, that discharge pressure (Pd) from the discharge chamber (10) is applied between the first valve element (101) and the piston (103), that suction pressure (Ps) from the suction chamber (9) is applied between the second valve element (102) and the piston (106), and that pressure (Pc) from the pressure-regulating chamber (1) is applied to portions of the directly abutting valve elements (101, 102).

17. The displacement control valve as in claim 16, **characterized by** first and second communication holes (105, 108; 134, 116) for introducing the pressure (Pc) from the pressure-regulating chamber (1) into back pressure chambers of the pistons (103, 106).

18. The displacement control valve as in claim 17, **characterized in that** the first communication hole (105) extends along the axis of the first valve element (101), the shaft (104), and the piston (103), integrally formed with each other, that the second communication hole (108) extends along the axis of the second valve element (102), the shaft (107), and the piston (106), integrally formed with each other, and that the portions of the abutting valve elements (101, 102) communicate with the first and second communication holes (105, 108), preferably via a stepped abutment face of at least one of the valve elements.

19. The displacement control valve as in claim 18, **characterized in that** the first communication hole (134) extends through a first body (110) providing a valve seat (109) for the first valve element (101), and that the second communication hole (116) extends through a second body (113) providing a valve seat (114) for the second valve element (102).

20. The displacement control valve as in claim 18, **characterized in that** a body (113) providing a valve seat (114) for the second valve element (102) and a cylinder bore slidably guiding the piston (106) is integrally formed with a fixed core (117) of the solenoid section.

21. The displacement control valve as in claim 7, **characterized in that** without control current the second valve element between the pressure-regulating chamber (1) and the suction chamber (9) maintains

a closed state, while the first valve element between the discharge chamber (10) and the pressure-regulating chamber (1) maintains an open state with a maximum valve opening degree for controlling the operating displacement of the variable displacement compressor to a minimum, and that with maximum control current supplied to the solenoid section, the second valve element between the pressure-regulating chamber (1) and the suction chamber (9) is in an open state with maximum valve opening degree, while, the first valve element between the discharge chamber (10) and the pressure-regulating chamber (1) is in a closed state, for controlling the operating displacement of the variable displacement compressor to a maximum.

22. The displacement control valve as in claim 7, **characterized in that** the valve is implemented into a variable displacement compressor in a refrigeration cycle causing a refrigerant, preferably CO_2, to perform refrigerating operation in a supercritical region with the refrigerant temperature above a supercritical refrigerant temperature.

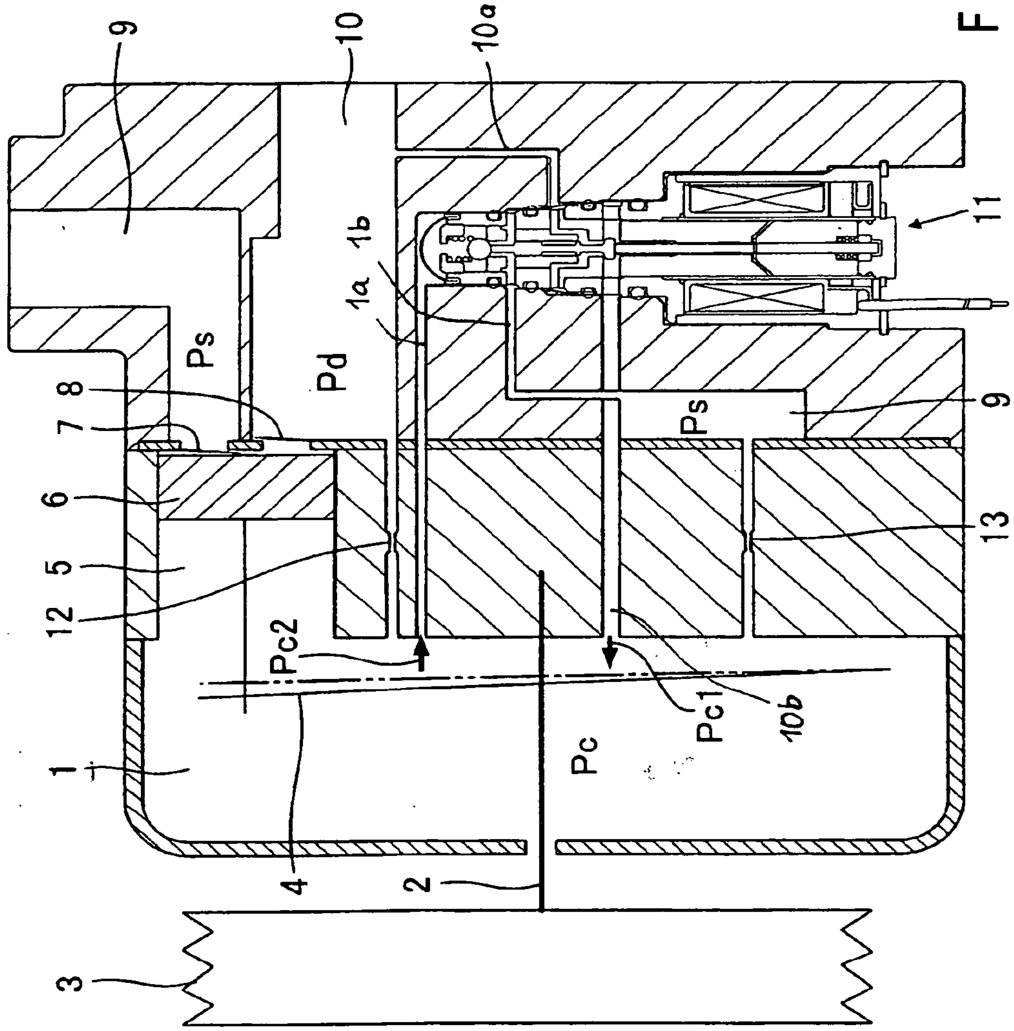

FIG. 1

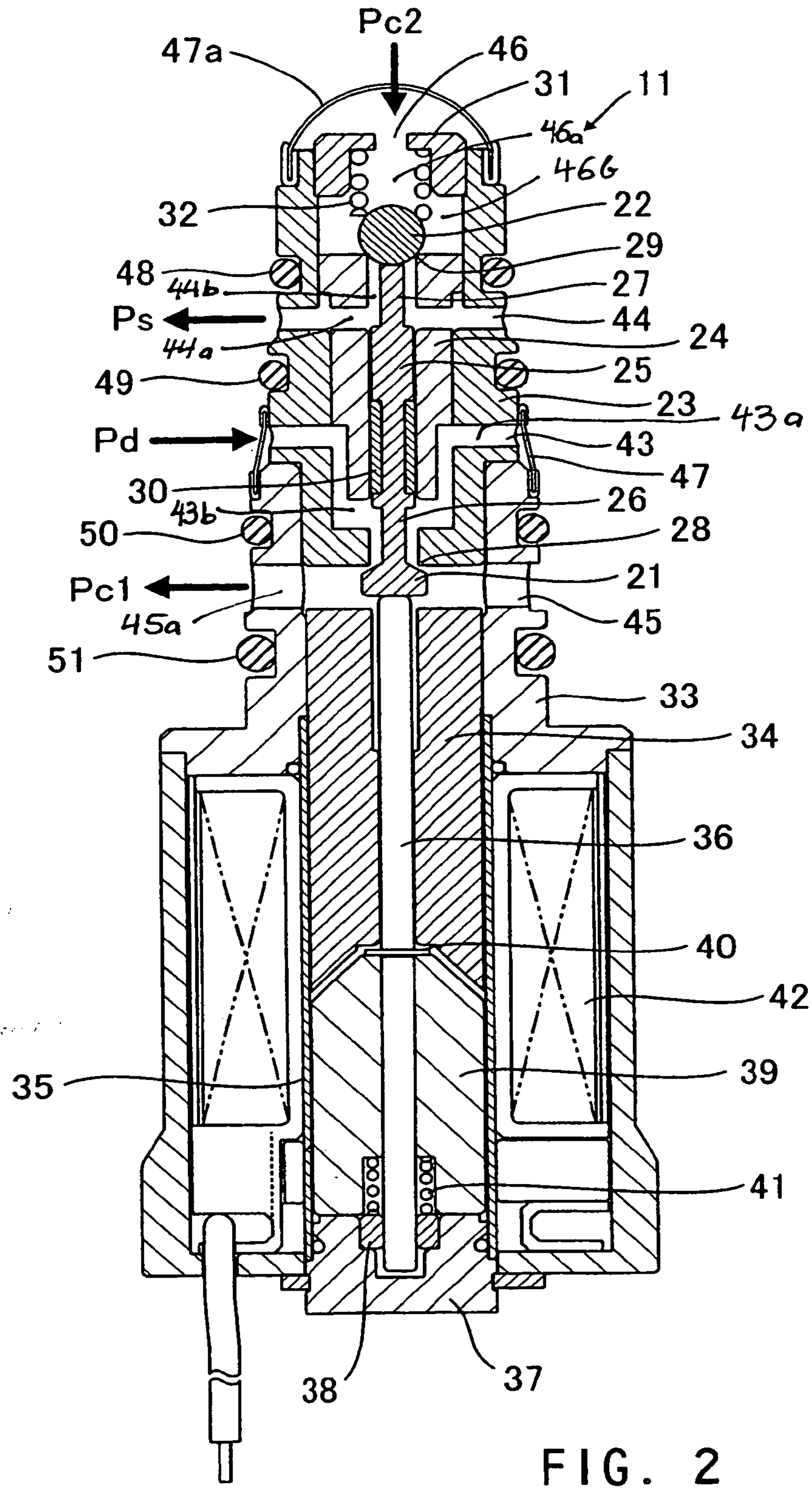
Pc2
47a
46
31
11
46a
46b
32
22
29
48
44b
27
Ps
44
24
44a
25
49
23
43a
Pd
43
47
30
43b
26
50
28
Pc1
21
45a
45
51
33
34
36
40
42
39
35
41
38
37

FIG. 2

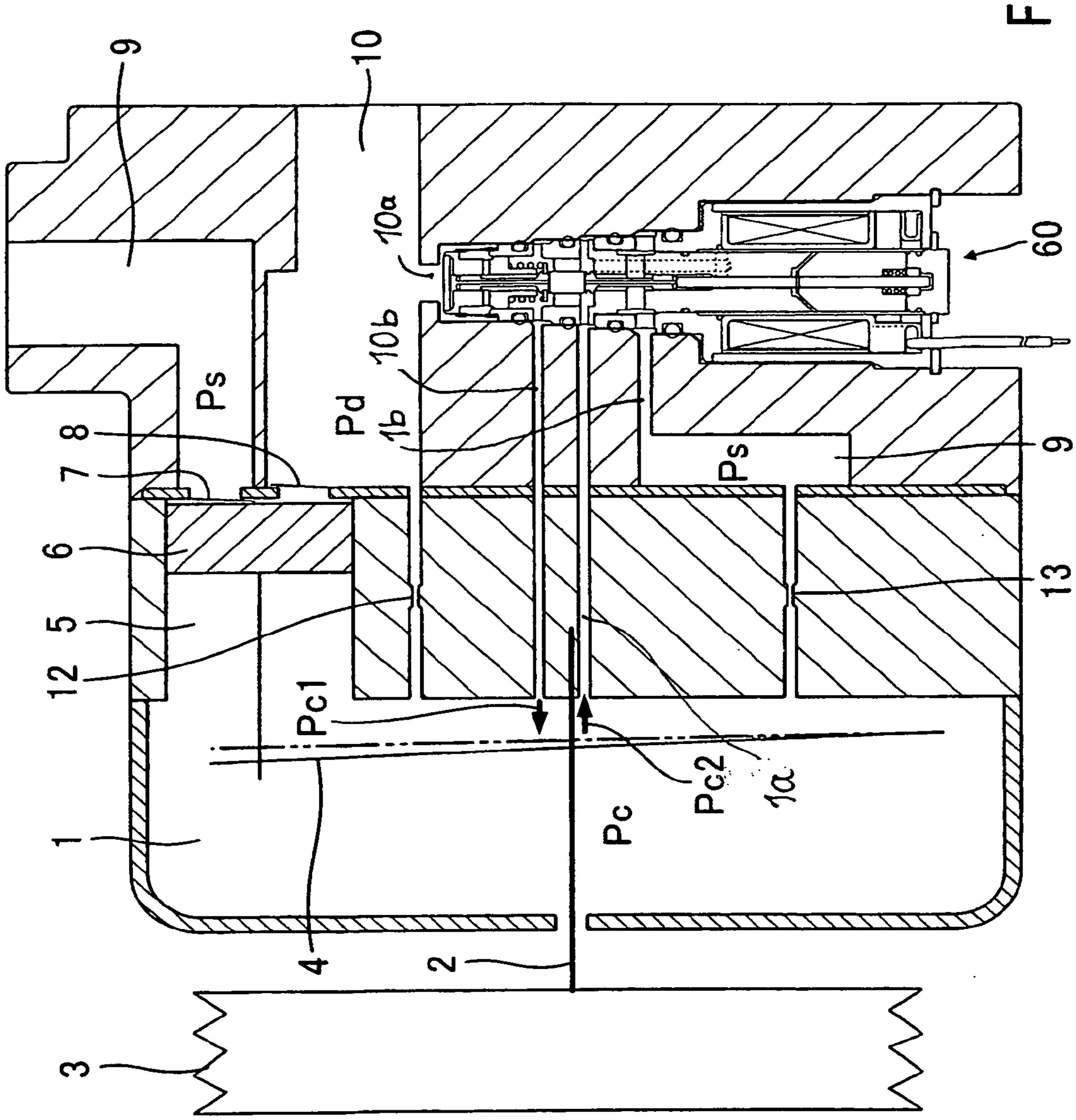

FIG. 3

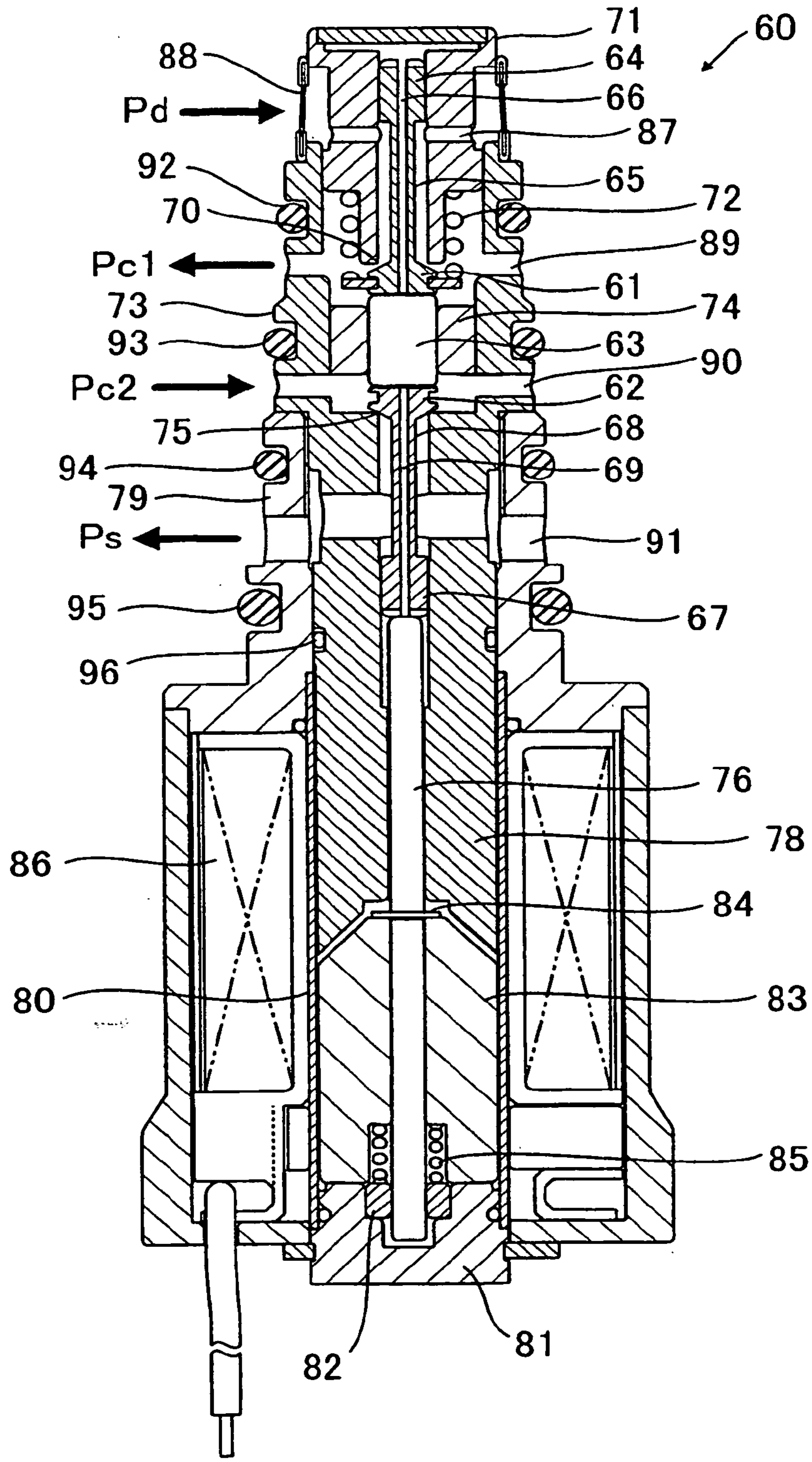
71
60
88
64
Pd
66
87
92
65
72
70
89
Pc1
61
73
74
93
63
90
Pc2
62
75
68
94
69
79
Ps
91
95
67
96
76
78
86
84
80
83
85
82
81

FIG. 4

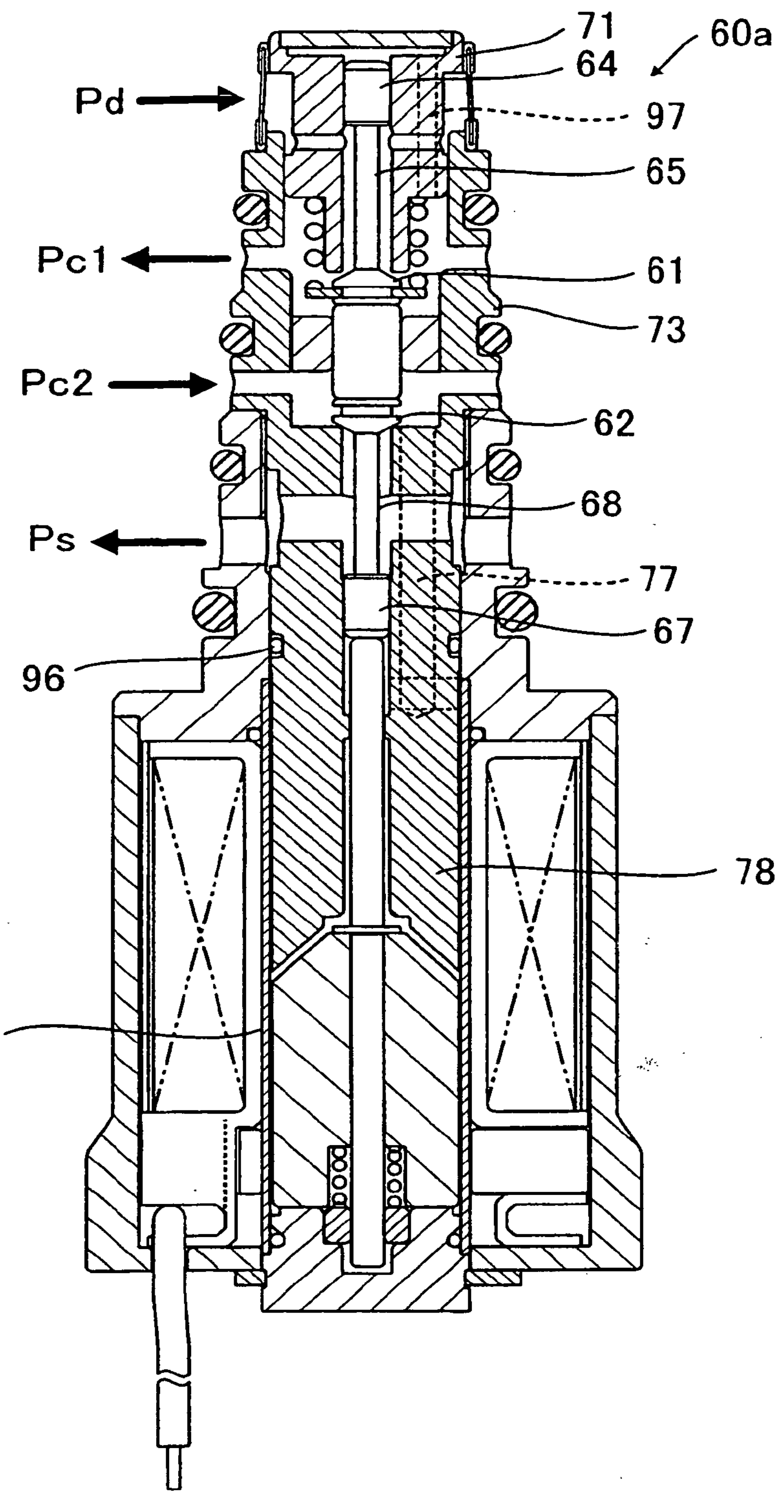

FIG. 5

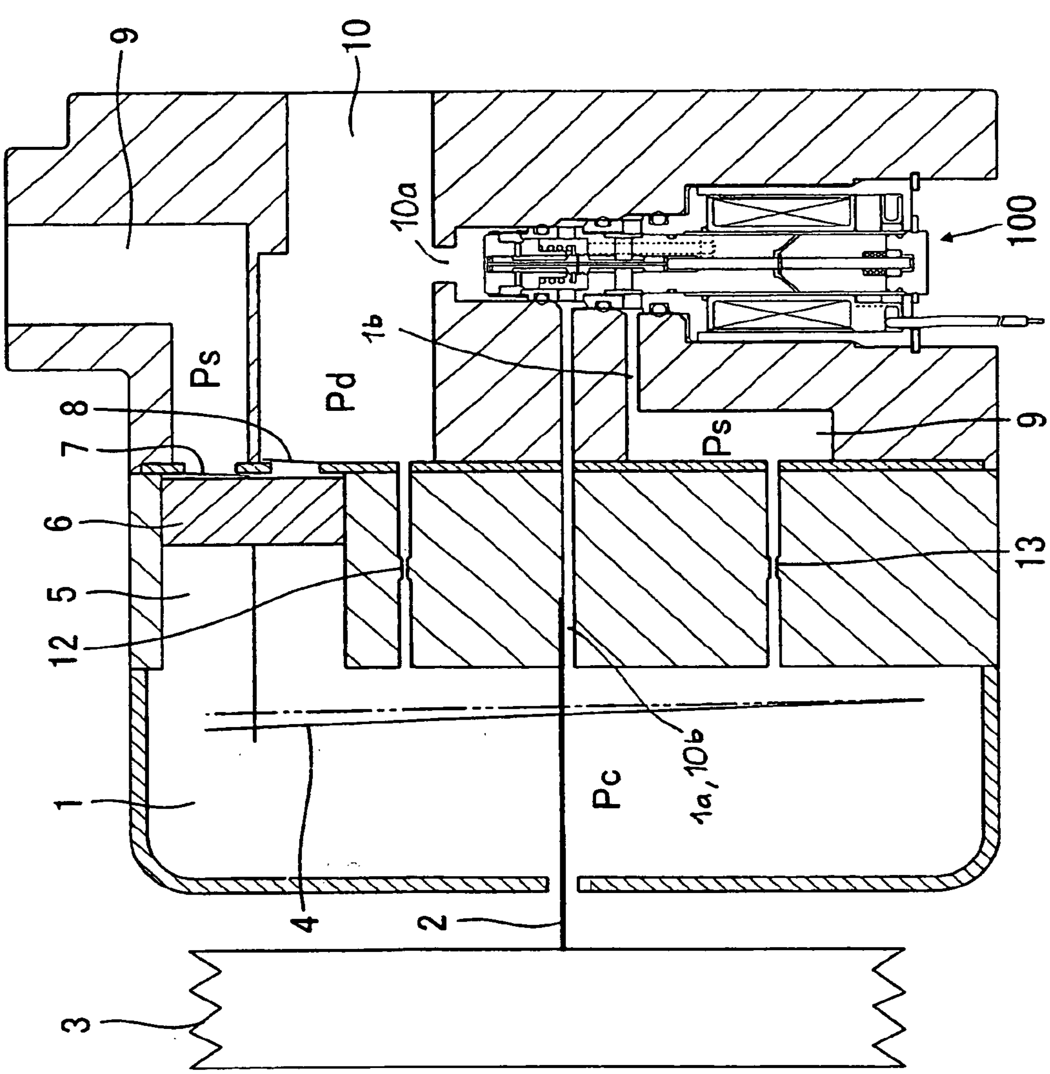

FIG. 6

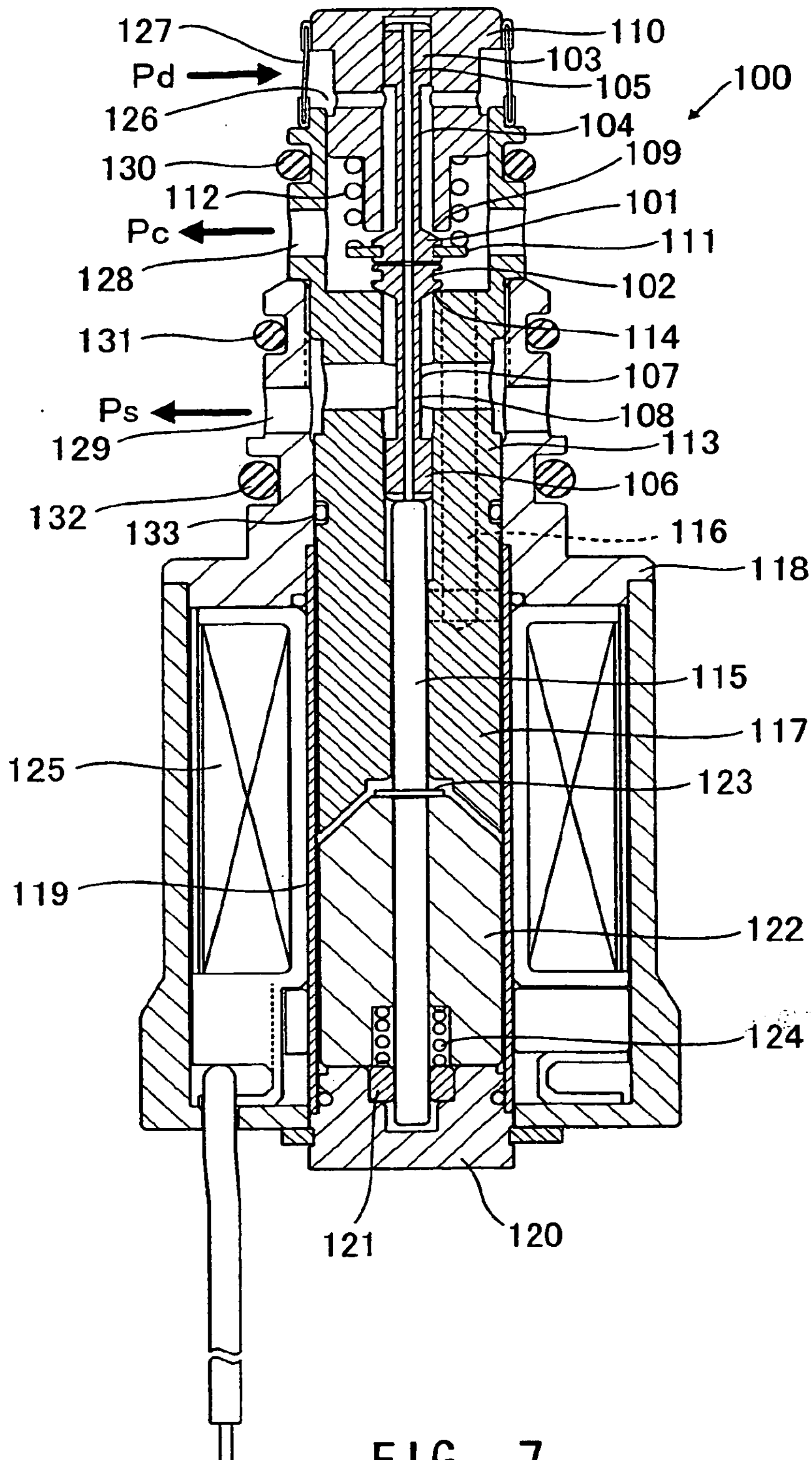
127
110
Pd
103
105
100
126
104
130
109
112
101
Pc
111
128
102
131
114
107
Ps
108
129
113
132
106
133
116
118
115
117
125
123
119
122
124
121
120

FIG. 7

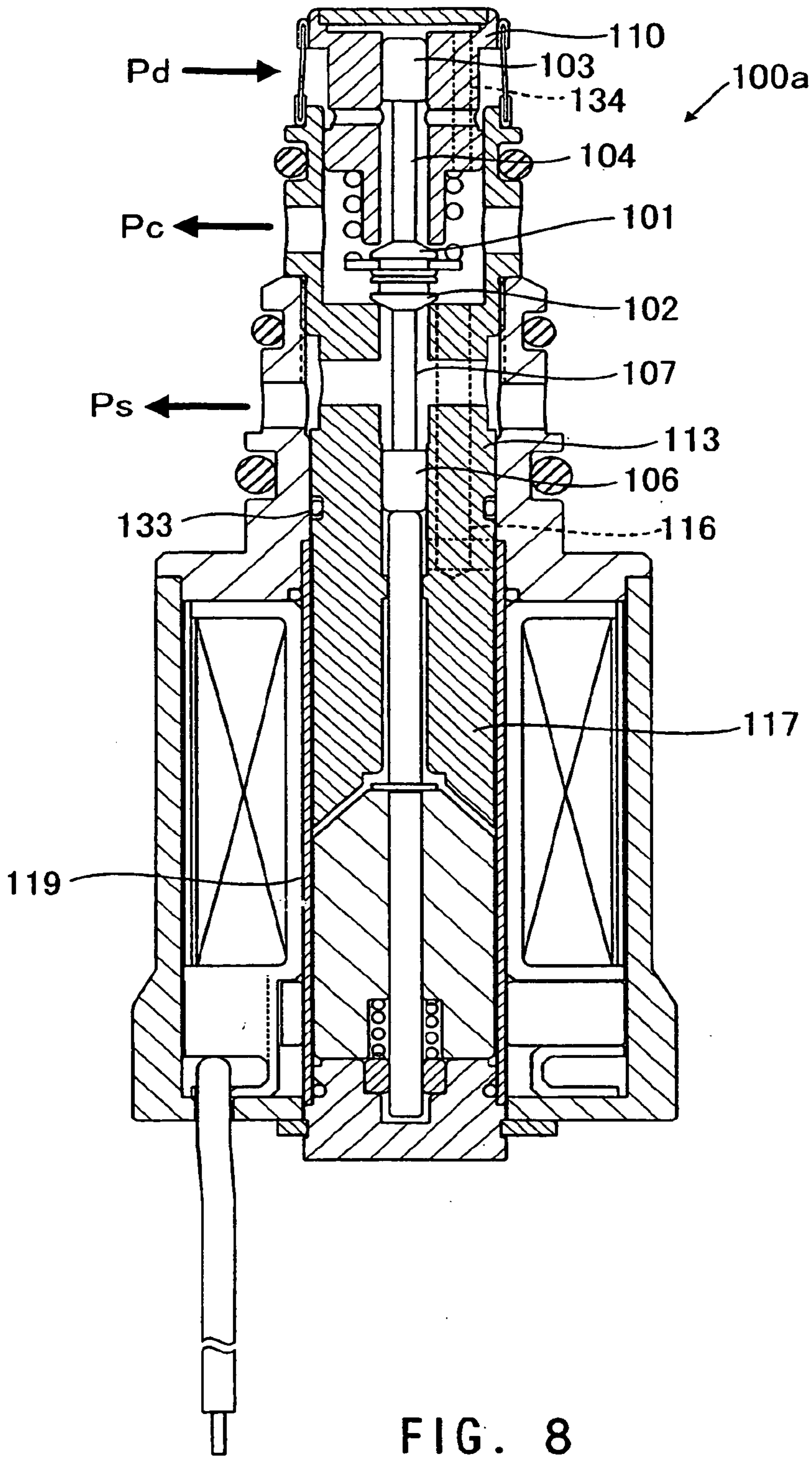

FIG. 8